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RESEARCH MEMORANDUM

COMPUTED TEMPERATURE DISTRIBUTION AND COOLING OF
SOLID GAS-TURBINE BLADES

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RESEARCH MEMORANDUM

COMPUTED TEMPERATURE DISTRIBUTION AND COOLING OF

SOLID GAS-TURBINE BLADES

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SUMMARY

Computations were made to determine the effects of gas temperature, blade-root temperature, blade thermal conductivity, and net gas-to-metal heat-transfer coefficient on the temperature distribution in a typical solid turbine blade. The computations covered a range of gas temperatures from 1500° to 2500° F, blade-root temperatures from 100° to 1000° F, blade thermal conductivities from 8 to 220 Btu/(hr)(sq ft)(°F/ft), and net gas-to-metal heat-transfer coefficients from 75 to 250 Btu/(hr)(sq ft)(°F).

The computations show that for turbine blades having a thermal conductivity approximating that of Inconel and stainless steel some improvement in the strength of the root section could be achieved by cooling the root and applying a ceramic coating to the blade section near the root; the effectiveness of these methods is negligible in cooling the upper half of the blade and other cooling methods are required. The effectiveness of reducing root temperature and of applying ceramic coatings in cooling the blade is improved by increasing the thermal conductivity of the blade material; however, very large increases in thermal conductivity (above that of Inconel or stainless steel) are required for these cooling methods to have an appreciable effect on the temperature of the upper half of the blade.

INTRODUCTION

The power output and the efficiency of gas turbines and jet-propulsion units are dependent on the temperature of the gases entering the turbine-blade passages. Because of the decreased strength and corrosion resistance of blade materials at high temperature, the maximum gas temperature is, at present, limited to about 1500° F. Cooling the blades would permit higher gas temperatures and hence higher efficiencies and outputs.

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Four methods of cooling turbine blades have been suggested:

- (1) Passing coolant (either gas or liquid) through passages in turbine blades
- (2) Cooling blade roots
- (3) Increasing blade thermal conductivity
- (4) Applying ceramic coating to blades

Hollow turbine blades have been used (reference 1) that permitted higher gas temperatures. No quantitative results, however, are available on this or the other methods of cooling.

As part of a general investigation of gas-turbine-blade cooling conducted at the NACA Cleveland laboratory, the results of computations are presented on the amount of cooling available by the last three of these methods. The results of these computations show the computed effect on the turbine-blade temperature of the following variables: (a) turbine-blade-root temperature, (b) thermal conductivity of blade material, (c) net gas-to-metal heat-transfer coefficient, and (d) gas temperature.

METHOD OF ANALYSIS

The effects of gas temperature, root temperature, blade thermal conductivity, and net gas-to-metal heat-transfer coefficient are shown by the computed results of the temperature distribution in a simulated typical turbine blade under several conditions.

A wedge-shaped turbine blade, having the dimensional characteristics shown in figure 1, was chosen for purposes of calculations.

The temperature at any point in the turbine blade was calculated by the equation developed in reference 2 for temperature distribution in a wedge-shaped blade. The equation of reference 2 (presented here in the form and notation used by Boelter, Cherry, and Johnson in a summarization of heat-transfer notes published by the University of California Press) for the temperature at any height in a wedge-shaped blade is

$$\frac{\tau_{\infty} - t}{\tau_{\infty} - t_0} = \frac{H_1(iu_E) J_0(iu) - J_1(iu_E) H_0(iu)}{H_1(iu_E) J_0(iu_0) - J_1(iu_E) H_0(iu_0)} \quad (1)$$

where

- T_w gas temperature, $^{\circ}\text{F}$
- t temperature of blade at height x above root, $^{\circ}\text{F}$
- t_0 blade-root temperature, $^{\circ}\text{F}$
- $H_0(iu)$ Hankel function of zero order
- $H_1(iu)$ Hankel function of first order
- i imaginary value, $\sqrt{-1}$
- $J_0(iu)$ Bessel function of first kind, zero order
- $J_1(iu)$ Bessel function of first kind, first order
- u function equal to $2b \left[L + \frac{\delta_E}{2} - x + \frac{\delta_E (1 - \tan \alpha)}{2 \tan \alpha} \right]^{1/2}$
- u_E function equal to $2b \left[\frac{\delta_E (1 - \tan \alpha)}{2 \tan \alpha} \right]^{1/2}$
- u_0 function equal to $2b \left[L + \frac{\delta_E}{2} + \frac{\delta_E (1 - \tan \alpha)}{2 \tan \alpha} \right]^{1/2}$
- b function equal to $(h_m/k_m \sin \alpha)^{1/2}$
- L height of blade, feet
- δ_E thickness of blade at tip, feet
- x distance along the blade from root, feet
- α angle between one side and center plane of blade
- h_m net gas-to-metal heat-transfer coefficient, $\text{Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$

The blade temperature at any point is dependent on the distance along the blade from the root, the angle between one side and the center of the blade, the thermal conductivity of the blade material, the gas temperature, the root temperature, and the net gas-to-metal heat-transfer coefficient. The net gas-to-metal heat-transfer coefficient as defined herein includes the resistance of any insulating coating applied to the blade surface. In the derivation of equation (1), a constant gas-to-metal heat-transfer coefficient, a constant blade thermal conductivity, and a constant gas temperature were assumed.

The gas velocity, and therefore the heat-transfer coefficient, varies somewhat from base to tip in actual operation; this velocity variation is approximately ± 10 percent of the average value and the assumption of a fixed value should give a good approximation of the temperature distribution. The variation in the thermal conductivity of the turbine blade material with temperature is also small (approximately 10 percent) for the range of temperatures involved and should have little effect on the distribution. In the impulse-type turbine, the chief gas-temperature drop occurs across the nozzle box; the gas in the blade passages remains at an essentially constant temperature.

The heat-transfer coefficient of the gas film for determining h_m used in the term b of equation (1) was calculated from results obtained by the Heat Transfer Section of the General Engineering Laboratory of the General Electric Company (GE Data Folder No. 71248). The General Electric data were correlated by the relation

$$\frac{h}{c_p G} \left(\frac{c_p \mu}{k} \right)^{2/3} = 0.198 \left(\frac{\psi G}{\mu} \right)^{-0.321} \quad (2)$$

where

- c_p specific heat of gas at constant pressure, Btu/(lb)(°F)
- G mass velocity of gas, (lb)/(hr)(sq ft)
- h heat-transfer coefficient of the gas film, Btu/(hr)(sq ft)(°F)
- k thermal conductivity of gas, Btu/(hr)(sq ft)(°F/ft)
- μ absolute viscosity of gas, (lb)/(hr)(ft)
- ψ mean perimeter of blade, feet

In the calculation of the heat-transfer coefficient, the flow of gas over the turbine blade was assumed to be parallel to the base; that is, having no vertical velocity component. If the gas flow were other than parallel to the base, the value of ψ in equation (2) would be slightly increased. Because h is inversely proportional to the 0.321 power of ψ , the change in h (caused by a small change in ψ) would be slight. It was also assumed that the heat transfer by radiation is small compared with the total heat transferred.

For a gas temperature of 1500° F, a relative velocity between gas and blade of 1000 feet per second, and a gas density of 0.0810 pound per cubic foot, the heat-transfer coefficient of the gas film was calculated as 250 Btu/(hr)(sq ft)(°F) by the use of equation (2). Because the exponent of h_m is 0.5 in the term b of equation (1), the error introduced by neglecting the change in the heat-transfer coefficient due to physical property changes with temperature would be slight, and the constant value of 250 Btu/(hr)(sq ft)(°F) was used at all gas temperatures.

Computations are presented for a range of gas temperatures from 1500° to 2500° F and root temperatures from 100° to 1000° F for several combinations of the following conditions:

Thermal conductivity of blade material [Btu/(hr)(sq ft)(°F/ft)]	Approximate metal equivalent
8	Inconel or stainless steel
25	Mild steel
220	Copper

Net gas-to-metal heat-transfer coefficient [Btu/(hr)(sq ft)(°F)]	Basic assumptions			
	Gas velocity relative to blade (ft/sec)	Gas density (lb/cu ft)	Ceramic-coating thickness (in.)	
			¹ k _c = 0.3	¹ k _c = 0.6
250	1000	0.0810	0	0
150	1000	.0810	.0096	.0192
75	1000	.0810	.0336	.0672

¹The symbol k_c refers to the thermal conductivity of the ceramic coating in Btu/(hr)(sq ft)(°F/ft).

RESULTS AND DISCUSSION

The computed temperatures at various points in the turbine blade are given in table I and are shown graphically in figures 2 to 5. Under all conditions the maximum temperature was found to exist at the tip of the blade.

The effect of gas temperature on the temperature distribution is shown in figure 2. It is assumed that the low blade-root temperature could be obtained by some form of root cooling. As shown by the figure, the upper two-thirds of the blade is substantially at the gas temperature in all cases.

In figure 3 the effect of root temperature on the temperature distribution is shown. Here again the upper part of the blade is at the gas temperature and is not appreciably affected by the root temperature. The temperature of the lower part of the blade (where the stresses are high) is considerably decreased when the root temperature is reduced to 100° F.

The effect of the turbine-blade thermal conductivity on the temperature distribution is shown in figure 4. An appreciable reduction in blade temperature is shown for very large increases in blade thermal conductivity above 8 Btu/(hr)(sq ft)($^{\circ}$ F/ft) (corresponding to Inconel or stainless steel) indicating that a study should be made of the thermal conductivities of heat-resistant materials to determine possible superiority over Inconel or stainless steel in this respect.

Figure 5 shows the effect of net gas-to-metal heat-transfer coefficient. As shown in the table of assumed conditions, the net heat-transfer coefficient of 250 Btu/(hr)(sq ft)($^{\circ}$ F) is obtained with a gas velocity of 1000 feet per second and a gas density of 0.0810 pound per cubic foot with no ceramic coating on the blade. This net heat-transfer coefficient is reduced to 150 and 75 Btu/(hr)(sq ft)($^{\circ}$ F) by the addition of various thicknesses of ceramic coatings. It is noted that with a blade thermal conductivity of 8 Btu/(hr)(sq ft)($^{\circ}$ F/ft) the addition of an insulating coating has a negligible effect but has an appreciable effect with a blade thermal conductivity of 220 Btu/(hr)(sq ft)($^{\circ}$ F/ft). There appears to be no gain in blade cooling by applying an insulating coating to the upper half of the blade in the case of the smaller blade thermal conductivity.

Figure 5 also indicates the increased effect of the root temperature on the temperature of the turbine blade in the case of the

higher blade thermal conductivity. It is apparent that root cooling is highly advantageous when the blade thermal conductivity is large.

CONCLUSIONS

It may be concluded, in general, that for solid turbine blades having a thermal conductivity similar to that of Inconel and stainless steel some improvement in the strength of the root section could be achieved by cooling the root and applying a ceramic coating to the blade section near the root. The effectiveness of these methods is negligible in cooling the upper half of the blade and other cooling methods are required. The effectiveness of reducing root temperature and of applying ceramic coatings in cooling the blade is improved by increasing the thermal conductivity of the blade; however, very large increases in thermal conductivity (above that of Inconel or stainless steel) are required in order for these cooling methods to have an appreciable effect on the temperature of the upper half of the blade.

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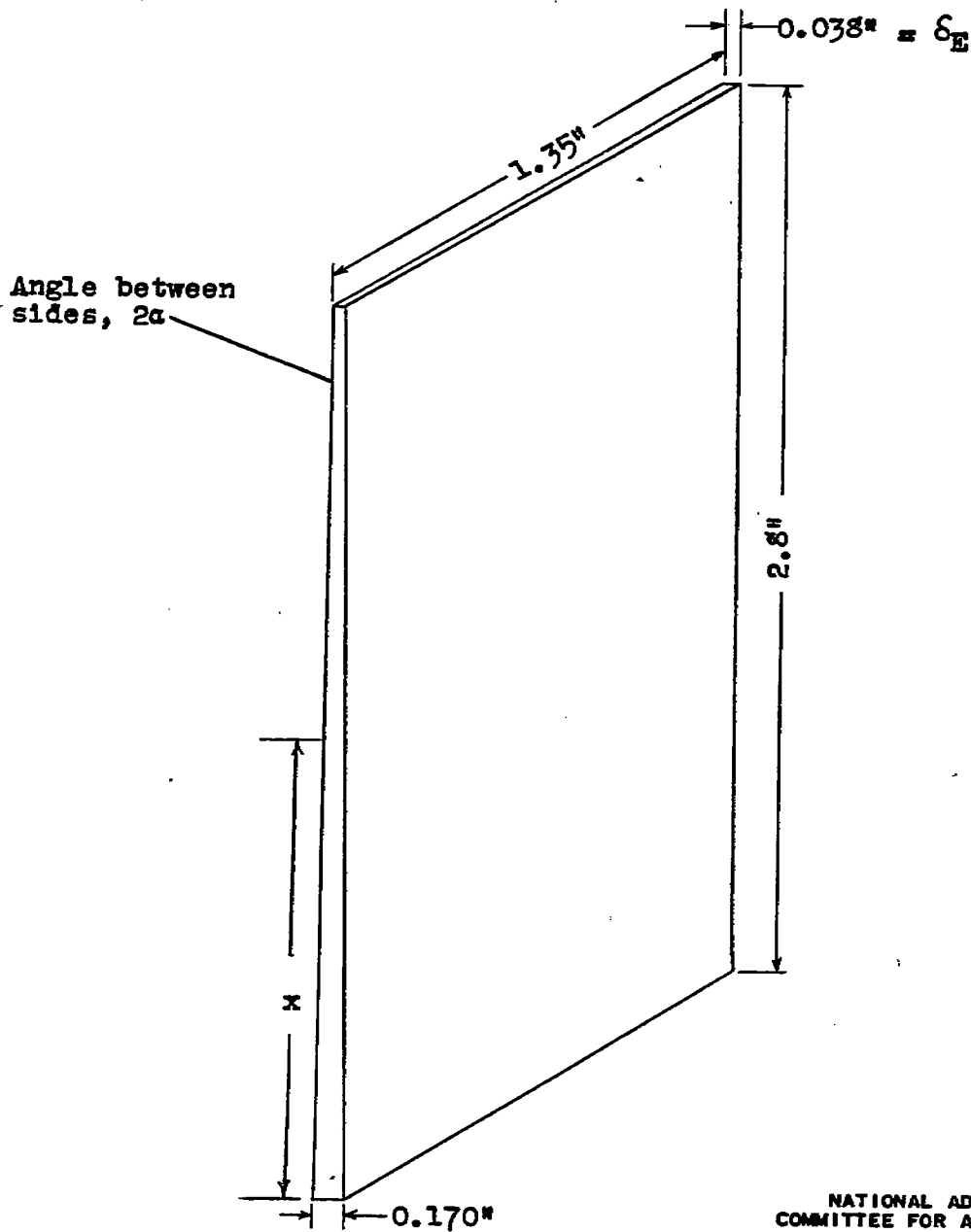
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1. Lorenzen, Christian: The Lorenzen Gas Turbine and Supercharger for Gasoline and Diesel Engines. Mech. Eng., vol. 52, no. 7, July 1930, pp. 665-672.
2. Harper, D. R., and Brown, W. B.: Mathematical Equations for Heat Conduction in the Fins of Air-Cooled Engines. NACA Rep. No. 158, 1922.

TABLE I - COMPUTED BLADE TEMPERATURES

Gas temperature, T_a (°F)	Blade-root temperature, t_o (°F)	Thermal conductivity of blade material, k_m [Btu/(hr) (sq ft)(°F/ft)]	Net gas-to-metal heat-transfer coefficient, h [Btu/(hr) (sq ft)(°F)]	Blade temperature, t (°F)				
				Distance from blade root, x (in.)				
				0.1	0.5	1.0	2.0	2.8 (tip)
1500	500	8	250	889	1443	1497	1500	1500
2000				1083	1914	1996	2000	2000
2500				1279	2386	2494	2500	2500
1500	100	8	250	645	1420	1496	1500	1500
	1000			1194	1471	1498	1500	1500
1500	500	25	250	756	1300	1464	1499	1500
		220		589	894	1151	1398	1454
1500	500	8	75	754	1290	1460	1492	1500
			150	887	1395	1489	1500	1500
1500	500	220	75	551	725	907	1156	1241
			150	576	826	1049	1314	1391
1500	100	220	75	171	415	670	1018	1137
			150	206	556	869	1239	1347
			250	225	652	1011	1357	1436

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Figure 1. - Wedge-shaped blade used in computation of temperature distribution.

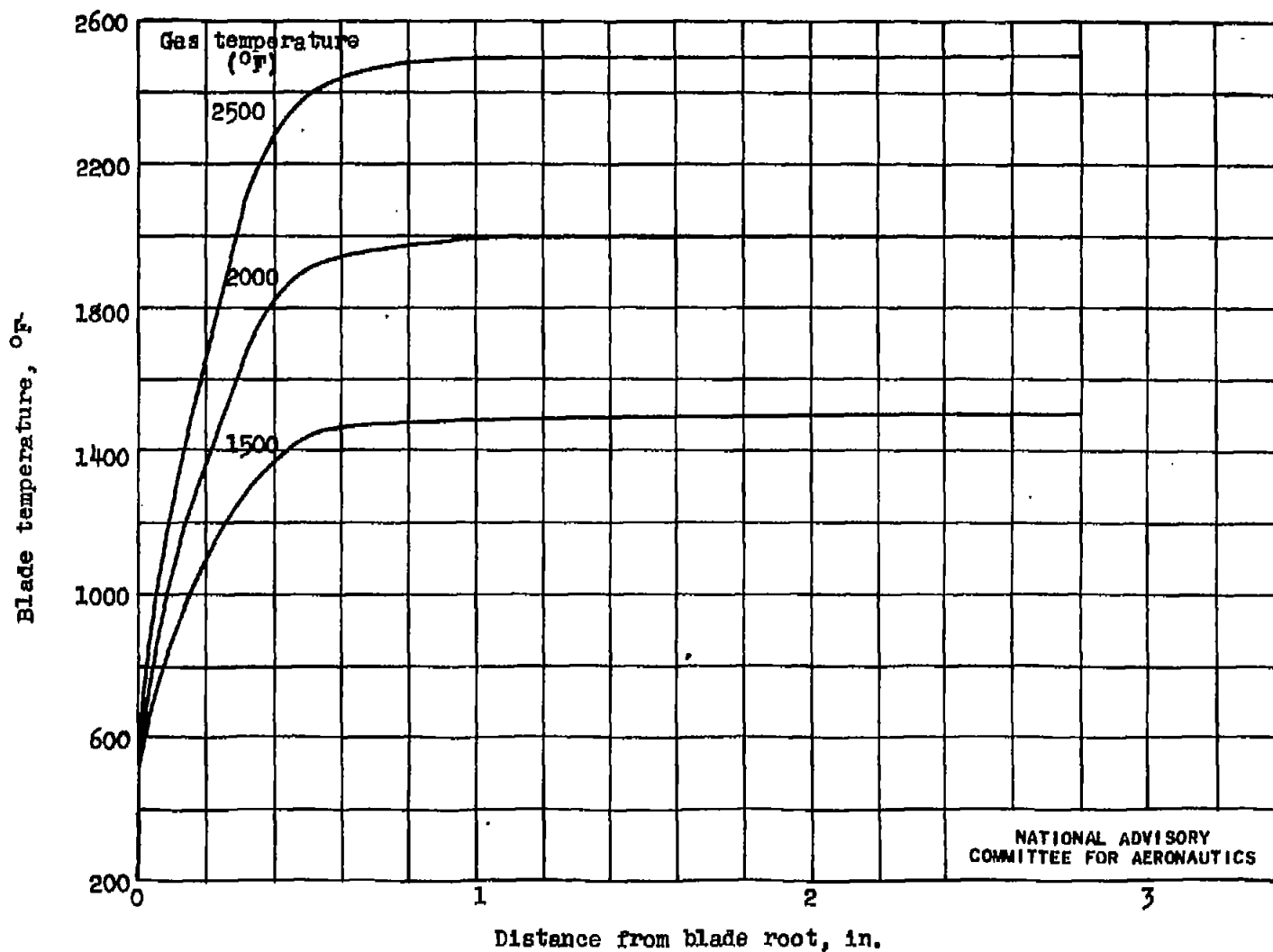


Figure 2.- Effect of gas temperature on temperature distribution in turbine blade. Root temperature, 500° F; blade thermal conductivity, 8 Btu/(hr)(sq ft)(°F/ft); net heat-transfer coefficient, 250 Btu/(hr)(sq ft)(°F).

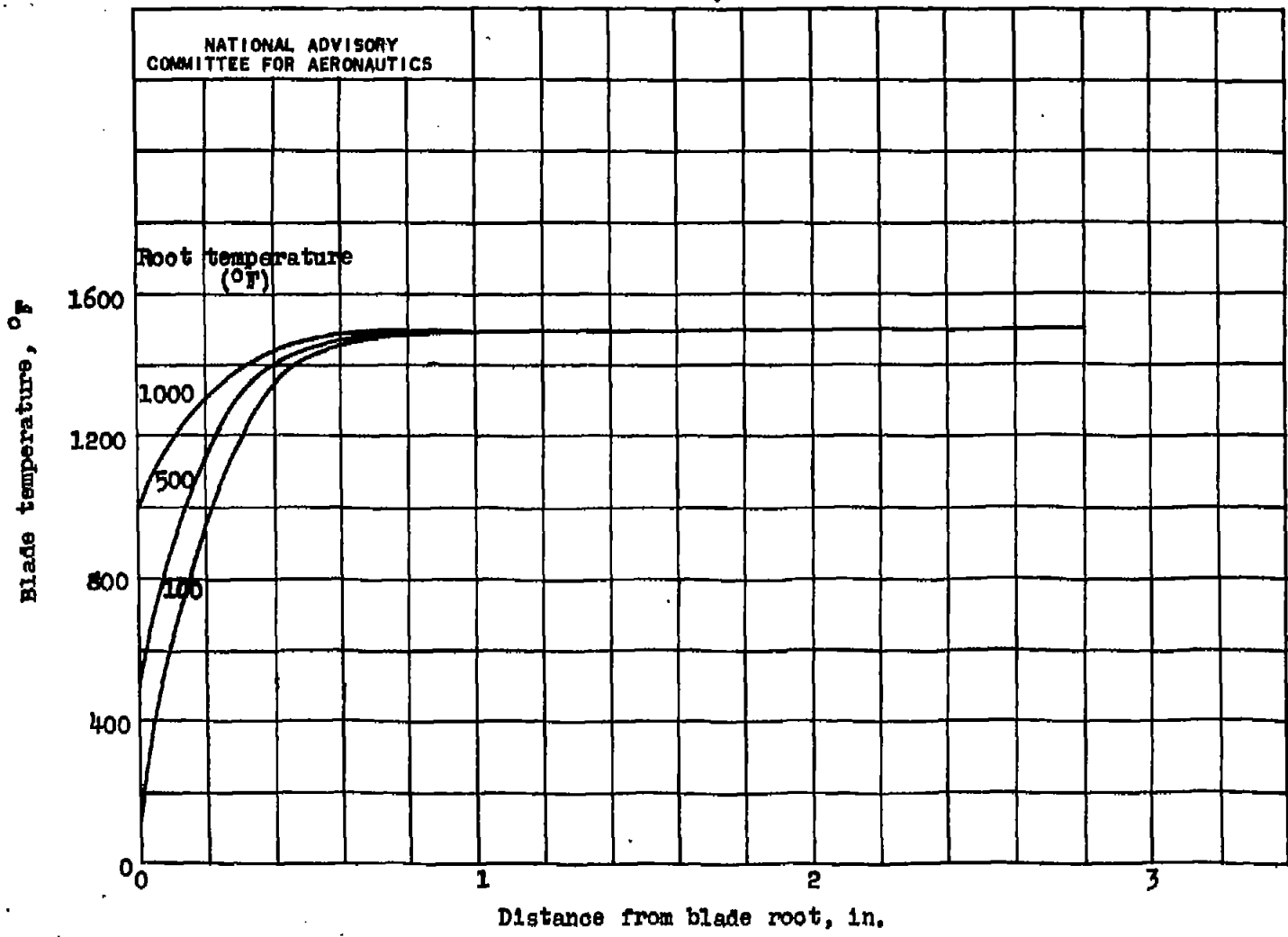


Figure 3.- Effect of root temperature on temperature distribution in turbine blade. Gas temperature, 1500° F; blade thermal conductivity, 8 Btu/(hr)(sq ft)(°F/ft); net heat-transfer coefficient, 250 Btu/(hr)(sq ft)(°F).

FIG. 3

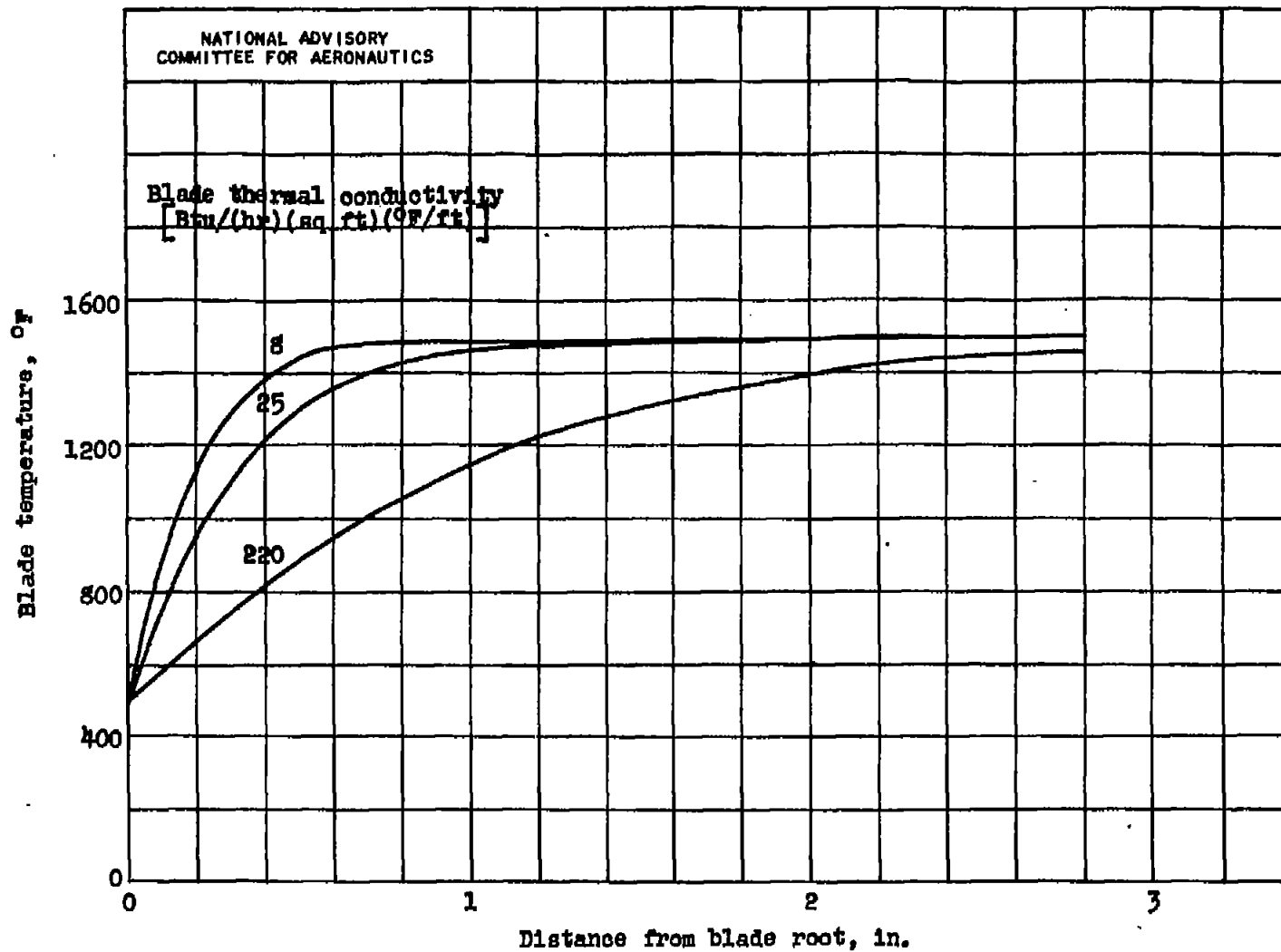


Figure 4.- Effect of blade thermal conductivity on temperature distribution in turbine blade. Gas temperature, 1500°F ; root temperature, 500°F ; net heat-transfer coefficient, $250\text{ Btu}/(\text{hr})(\text{sq ft})(^{\circ}\text{F})$.

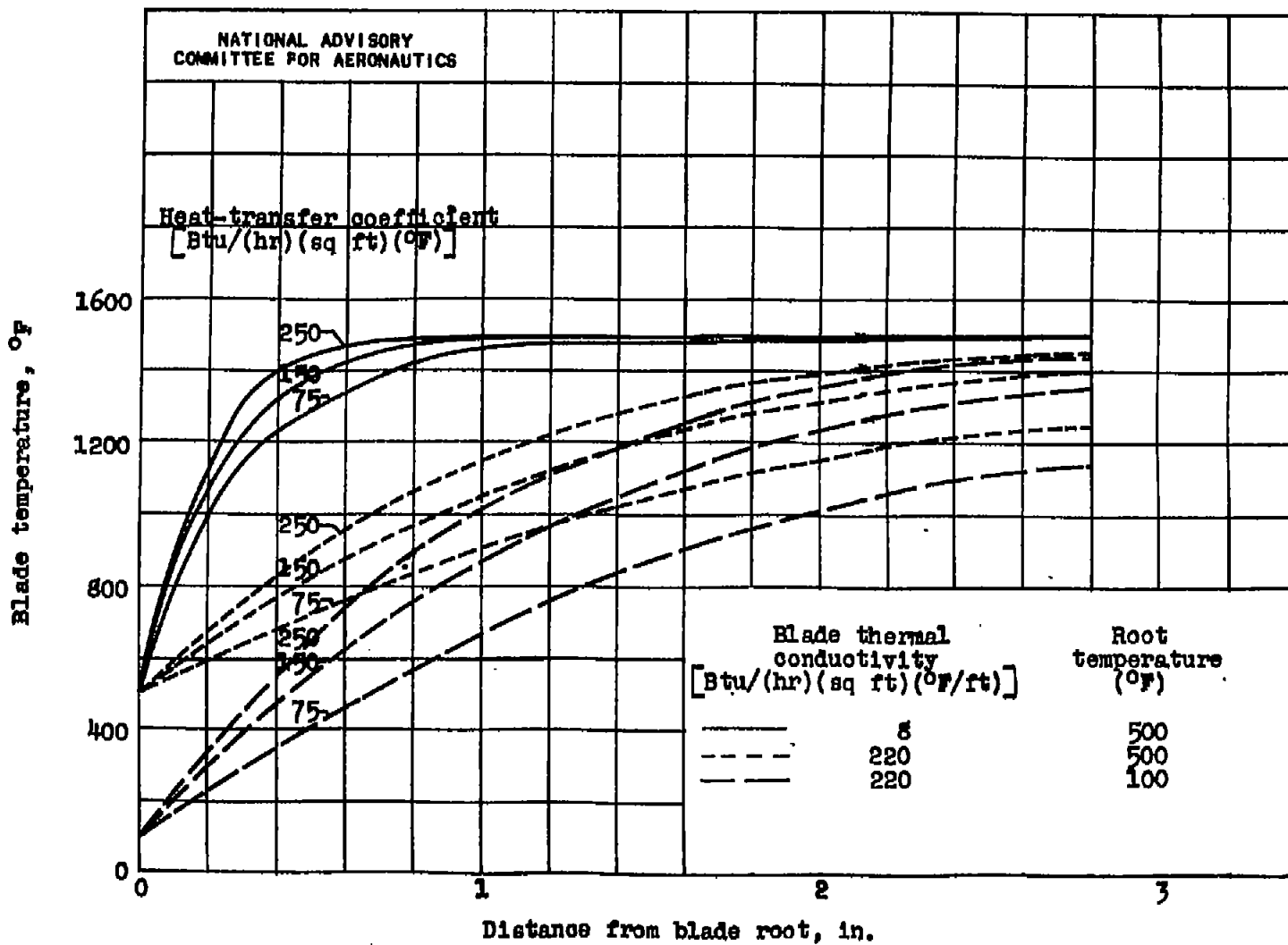


Figure 5. - Comparison of net heat-transfer coefficient effects at different root temperatures and blade thermal conductivities. Gas temperature, 1500° F.

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